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DIESEL-ENGINE INVESTIGATIONS

IGNITION-CHAMBER ENGINES

By Kurt Neumann

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Many investigations have been made of the injection and combustion processes in air-injection and airless-injection engines. As much cannot be said, however, regarding ignition-chamber engines. Since this type of engine, on account of its structural advantages, is worthy of attention in these days of endeavor to develop the high-speed Diesel vehicle engine, it seemed desirable to make a thorough investigation of the functioning process and injection and combustion characteristics of ignition-chamber engines.

For these experiments, on account of the necessarily accurate measurements, only a stationary engine could be used. The experiments were therefore performed exclusively on an engine made for the institute by the Körting Brothers, of Hannover-Linden. This single-cylinder, four-stroke-cycle 18 hp engine has the following characteristics.

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*"Untersuchungen an der Dieselmaschine. VII. Die Vorkammermaschine." From Dieselmaschinen IV, pp. 75-82, published by the V.D.I. Publishing Company, Berlin, 1929. Abstract of a report which was published in full in Forschungsheft No. 309.

Stroke	$s = 316$ mm	
Cylinder diameter	$d = 190$ "	
Normal revolution speed	$n = 350$ r.p.m.	
Cylinder volume	$V_h = 8.960\,l$,	$\epsilon_h = 1$
Compression space	$V_k = 0.444\,l$,	$\epsilon_k = 0.0495$
Chamber volume	$V_z = 0.195\,l$,	$\epsilon_z \doteq 0.0218$

The functioning of this engine differs considerably from that of other ignition-chamber engines. The passage between the ignition chamber and the cylinder has a small diameter, in order to cause a great pressure drop during the compression and a consequent high velocity of the air flow into the ignition chamber (V.D.L., Vol. 70 (1926), p. 771, Fig. 13). Shortly before the dead center, the fuel is injected into the passage in such a way that it impinges on the highly cooled wall of the passage and dissipates its kinetic energy. The fuel spray is further atomized and driven back by the inrushing air into the ignition chamber, where it ignites and burns in an excess of air.

A high-pressure increase is thus suddenly produced in the chamber. The flow is reversed and the principal combustion takes place in the cylinder aided by the energy of the inflowing air and the further injection of fuel.

In the conversion of the chemical energy of the fuel into heat and useful work, the time available for the starting and completion of the combustion is of decisive importance. In an

air-injection Diesel engine, the injection air serves as the energy carrier and produces adequate atomization and combustion of the injected fuel.

Since the flow produced by the injection air is, in a certain sense, replaced in the experimental engine by flows which owe their origin not to external agencies but to the interaction between the cylinder and ignition chamber, I am presenting, before investigating the normal functioning of the engine under full load, an analysis of the flow in the cylinder and ignition chamber, in which the revolution speed appears as a variable parameter and the effect of the combustion is temporarily eliminated.

The Flow in the Cylinder and Ignition Chamber

The injection energy of the liquid fuel is small in an ignition-chamber engine, due to the relatively low pressure of the pump. Hence the flow is here of especial importance. Since the injection occurs before the dead center of the piston, the flow during the compression is considered first.

If V_k, V_z, and V_h denote respectively the volumes of the compression space, ignition chamber and cylinder (Fig. 1), two compression ratios, $\epsilon_k = V_k : V_h$ and $\epsilon_z = V_z : V_h$ must then be considered in the case of the ignition-chamber engine. Due to the relatively small cross section f of the connecting passage, the pressure p_1 in the cylinder rises more rapidly during the compression than the pressure p_2 in the chamber.

The higher pressure in the cylinder produces a flow in the passage which is continued in the chamber. The pressure variations in the cylinder and chamber depend on the cross section f and on the piston speed or revolution speed of the engine. If the latter is very low, the pressures in the cylinder and chamber are quickly equalized, and the final compression pressure corresponds to the compression ratio $\epsilon = \epsilon_k + \epsilon_z$. At very high revolution speeds (limiting case $n = \infty$) the final compression pressure is determined only by ϵ_k, since the compressed air in the cylinder does not have time to flow into the chamber.

Figure 2 shows the pressure variation during the compression. The pressure at the beginning of the compression is 1 atm. abs. (absolute atmosphere), the compression taking place according to the law $PV^{1.35}$ = constant. Three limiting pressure curves are obtained. If equal pressures $p_2 = p_1$ prevail in the cylinder and chamber, then $p'' = f(\alpha)$ yields the final pressure $p_k = 38.7$ atm. abs. For infinitely quick compression the cylinder pressure $p_1' = f(\alpha)$ and the final pressure $p_k = 61.5$ atm. abs., while the chamber pressure $p_2' =$ 1 atm. abs. remains unchanged.

For revolution speeds between $n = 0$ and $n = \infty$ corresponding to the actual performances, the cylinder pressure $p_1 = f(\alpha)$ is generally above and the chamber pressure $p_2 = f(\alpha)$ below the limiting pressure curve $p'' = f(\alpha)$, whereby the difference $\Delta p = p_1 - p_2$ is inversely proportional to

the cross section f and directly proportional to the revolution speed n. The velocity of the air flow from the cylinder into the chamber and, consequently, the turbulence in the chamber increase, as the value of Δp increases. The pressure in the cylinder and chamber is a function of $V_k : V_h$, $V_z : V_h$, and $f : n$. The criterion for the turbulence of the chamber contents is E, which depends on the weight and velocity of the air flowing into the chamber.

Thus, if the air mass $d\,G_2$ flows into the chamber with the velocity w_2 during the crank angle $d\,\alpha$,

$$G_2 = \frac{P_{10}^{\frac{m-1}{m}}}{R\,T_{10}}\,\frac{\mu f}{6n}\int_0^\alpha w_2\,P_2^{\frac{1}{m}}\,d\,\alpha\ \text{kg},$$

in which P_{10}, T_{10} refer to the initial condition in the cylinder at the beginning of the compression, the coefficient of discharge being μ and the velocity being

$$w_2 = 44.8\sqrt{T_{10}\left(\left|\frac{p_1}{p_o}\right|^{\frac{m-1}{m}} - \left|\frac{p_2}{p_o}\right|^{\frac{m-1}{m}}\right)}\ \text{m/s}$$

which is determined from the indicated pressure curves for the cylinder and the ignition chamber, respectively, $p_1 = f(\alpha)$ and $p_2 = f(\alpha)$. The momentary energy of flow is

$$d\,E_2 = d\,G_2\,\frac{{w_2}^2}{2\,g},$$

from which follows

$$E_2 = \frac{p_{10}^{\frac{m-1}{m}}}{2\ g\ R\ T_{10}} \frac{\mu f}{6n} \int_0^\alpha w_2^3\ p_2^{\frac{1}{m}}\ d\,\alpha\ \text{mkg}$$

Of special importance during the compression, is the variation in the energy of flow (turbulence in the chamber), $\frac{d\ E_2}{d\ \alpha}$ mkg per crank-angle degree, since it is desirable for the turbulence to be as great as possible during the injection shortly before the dead center of the piston.

For the determination of μ, the connecting passage between the cylinder and chamber was used as the outlet of a compressed-air tank, through which a measured quantity of air at a known pressure was allowed to escape into the open air. The value $\mu = 0.61$ was thus obtained. Figures 3-4 show, for a constant revolution speed of $n = 311$ r.p.m., the quantity of air G_2 flowing into the chamber, the flow velocity w_2, the flow energy E_2, and the variations $d\ G_2/d\alpha$ and $d\ E_2/d\alpha$ plotted against the crank angle α. It is seen that these quantities increase rapidly in the second half of the compression stroke and reach their maximum values 14° before the dead center. It is especially worth noting that the turbulence $dE_2/d\alpha$ is very great at this instant, but diminishes rapidly as the piston approaches the dead center.

Other conditions remaining the same, the flow characteristics and consequently the conditions for the ignition and precombustion vary as the revolution speed of the engine. The investigation of the engine at three different revolution speeds

gave the results represented in Figure 5. The maximum value of the air quantity $\int_0^{180} d\,G_2$ forced over during the compression was already exceeded at the normal r.p.m. of 350, while the turbulence $(dE_2/d\alpha)_{max}$ in the chamber during the injection was the greatest at this r.p.m. G_2 increases moderately with increasing r.p.m., while the turbulence diminishes rapidly.

The pressure difference $\Delta p = f(\alpha)$ and the velocity of flow $w_2 = f(\alpha)$ between the cylinder and chamber increase with increasing engine speed. The flow energy E_2 produced during the entire compression stroke reaches the same magnitude as in air-injection engines with highly compressed air. The turbulence in the ignition chamber, due to its smaller capacity, is greater than in the combustion space of an air-injection engine. Since there is sufficient fresh air in the chamber and the temperature is high enough, as will be shown later, all the conditions are fulfilled, with the proper injection of the fuel, for producing a quick and sure ignition and for starting the combustion in the chamber in such manner as to cause a high flow velocity from the chamber and thereby an efficient combustion in the cylinder for every compression stroke.

Investigation of the Functioning at Full Load

The engine was tested at nearly full load. The brake horsepower was absorbed by an electric eddy-current brake and the air used by the engine was measured by an air meter. The cooling

water was measured separately for the cylinder and ignition chamber. The wall temperatures of the ignition chamber and of the connecting passage were determined by thermocouples. Moreover, the mean temperature of the gas in the chamber was measured by a platinum-rhodium thermocouple. The pressure changes in the cylinder and chamber were shown by ordinary and offset indicator diagrams. A third indicator described the needle-lift diagram. Gas samples could be taken from the chamber and cylinder by means of electrically controlled valves at any desired crank angle. The tests gave the following values:

Brake horsepower	N_e	= 17.8 b.hp
Mean indicated piston pressure	p_i +	= 7.15 atm.
	p_i −	= 0.02 "
Engine speed	n	= 346 r.p.m.
Fuel consumption	B	= 4.32 kg/h
Air consumption (15°C, 1 atm.)	L_r	= 75.53 m³/h
Cooling water:		
Cylinder	K_k	= 285 kg/h
Chamber	K_z	= 27 "
Inlet temperature	t_e	= 8.2°C
Outlet temperatures:		
Cylinder	t_{ak}	= 67.3°C
Chamber	t_{az}	= 67.0°C
Exhaust temperature	t_z	= 436°C
Mean chamber temperature	t_{2m}	= 672°C
Wall temperatures:		
Ignition chamber	ϑ_1	= 95°C
	ϑ_2	= 86°C
Connecting passage	ϑ_1'	= 128°C
	ϑ_2'	= 101°C

Fuel, gas oil:		
Lower heating value	h_u	= 10070 kcal/kg
Theoretical air quantity (15°C, 1 atm.)	L_{min}	= 12.29 m³/kg
Gas components:		
Exhaust gases (at room temp.)	CO_2	= 0.106
	O_2	= 0.065
Chamber and combustion gases in cylinder according to crank angle α. See Figures 9 and 10.		
Test results:		
Indicated power	N_i	= 24.6 hp
Mechanical efficiency	η_m	= 0.724
Specific fuel consumption	B_i	= 0.176 kg/hp_i/h
	B_e	= 0.243 kg/hp_e/h
Specific air consumption (15°C, 1 atm.)	L_r	= 17.48 m³/kg
Mean effective piston pressure	p_e	= 5.17 atm.
Air-excess coefficient	λ	= 1.42
Thermal efficiency	η_{ti}	= 0.357
	η_{te}	= 0.259
Volumetric efficiency	η_λ	= 0.812

Heat balance in fractions of the heat supplied

Useful work	Q_e = 11250 kcal/h,	q_e = 0.259	q_i=0.358
Frictional work	Q_ρ = 4310 "	q_ρ = 0.099	
Heat in cooling water:			
Cylinder	Q_{kk} = 16860 "	q_{kk} = 0.388	q_k=0.424
Chamber	Q_{kz} = 1590 "	q_{kz} = 0.036	
Heat in exhaust gases	Q_z = 10700 "	q_z = 0.246	
Remainder	Q_r = -1210 "	q_r = -0.028	
Heat supplied	Bh_u = 43500 kcal/h	Σq = 1000	

Figure 6 shows the wall-temperature curves for the passage and ignition chamber. Heat transmitted to the cooling water in

fractions of the total amount:

Chamber	$Q' = 1594$ kcal/h,	$q_{chamber} = 0.60$
Passage	$Q'' = 1060$ " ,	$q_{passage} = 0.40$
	$\Sigma Q = 2654$ kcal/h,	$\Sigma q = 1.00$

The cooling water test for the chamber alone gave $Q_{kz} =$ 1590 kcal/h in close agreement with the above.

Processes in the Cylinder and Ignition Chamber

Calculation methods.- The cylinder and ignition chamber work together and are considerably affected by the form and working cycle of the engine. The temperature curve in the chamber $t_2 = f(\alpha)$ can be determined from the measured mean gas temperature in the ignition chamber $t_{2_m} = 672^\circ C$ and from the indicator pressure diagram of the chamber $p_2 = f(\alpha)$. This is

$$T_2 = \frac{T_{2_m}}{\left(p_2^{\frac{m-1}{m}}\right)_m} p_2^{\frac{m-1}{m}} \ ^\circ\text{abs.}$$

Figure 7 shows the pressure and temperature curves for the chamber.

The combustion in the chamber is determined by the composition of the charge at the instant of ignition. The injection process and the course of the combustion in the cylinder are necessarily affected by the pressure increase in the chamber resulting from the combustion. At the beginning of the compression, the chamber contains the quantity of air

$$G_{l_{20}} = \frac{P_{20}\ V_z}{R\ T_{20}} = 0.091 \text{ g.}$$

The quantity of air forced into the chamber, up to the crank angle α, is determined by $G_{l_2} = f(\alpha)$, whereby the temperature in the cylinder at the beginning of the compression is

$$T_{10} = \frac{(1 + \epsilon_k + \epsilon_z)\ P_{10}}{\frac{\epsilon_k\ P_{rk}}{T_{rk}} + \frac{\epsilon_z\ P_{r_2}}{T_r} + \frac{848\ L_r}{24.4 \times 30\ n\ V_h}} = 366^\circ \text{ abs.}$$

or $t_{10} = 93^\circ C$.

The turbulence in the chamber varies during the compression stroke. The periodic variation $dE_2/d\alpha$ is a function of the crank angle α. Hence the weight of the air $G_{l_{20}} + G_{l_2}$ in the chamber is fixed for every crank angle α. Since the instant of the ignition can be deduced from the offset indicator diagram of the chamber pressures p_2 (It is 4.5° before the piston dead center), the amount of the air in the chamber during the ignition is also determined. For $G_{l_2} = 1.74$ g it is

$$G_{l_{20}} + G_{l_2} = 1.83 \text{ g.}$$

The fuel present in the chamber at the same instant depends on the law of injection. The injection time z_e corresponds to the crank angle $\alpha_e - \alpha_a$. The fuel injected from α_a to any desired angle α is then

$$B = \frac{\mu\ f\ \gamma_b}{6\ n} \int_{\alpha_a}^{\alpha} w_b\ d\ \alpha \text{ kg,}$$

Hence the injection velocity, derived from the pressure differ-

ence between the pump and the cylinder, is

$$w_b = \sqrt{\frac{2\ g\ 10^4\ \Delta\ p}{\gamma_b}}\ m/s$$

and the discharge coefficient μ, derived from the quantity of fuel injected for each cycle, is

$$\mu = \frac{6\ n\ B_{total}}{f\ \gamma_b \int_{\alpha_a}^{\alpha_e} w_b\ d\ \alpha}$$

The mean injection velocity is 158 m/s and the corresponding discharge coefficient is $\mu = 0.64$. If the amount of injected fuel B_α is calculated for different injection angles $\alpha - \alpha_a$, it is found that the former increases directly as the latter. Hence, up to the beginning of the ignition in the chamber (4.5° before the dead center), $B_z = 0.086$ g of fuel is injected. The mixture in the chamber is therefore characterized by the excess-air coefficient

$$\lambda = \frac{L_z}{B_z\ L_{min}} = 1.46.$$

The combustion in the chamber accordingly occurs with considerable air excess. From the beginning of the injection to the instant of ignition, λ decreases from $\lambda_{-13^\circ} = \infty$ to $\lambda_{-4.5^\circ} = 1.46$ (Fig. 8).

The presence of much excess oxygen in the chamber is also known from the analysis of the gas samples taken through the

controlled valve. In these samples, no evidence of incompletely burned hydrogen and carbon monoxide was found. The course of the combustion in the ignition chamber can be followed by taking samples at different crank angles. This shows, in agreement with the indicator diagram, that the pressure increase in the chamber occurs only once during each cycle. The combustion is indeed completed, but does not reach its maximum value, which would correspond to the maximum CO_2 content of the charge. Carbon is liberated during the combustion.

Since the flow between the cylinder and chamber is reversed at the instant of ignition, due to the pressure rise in the chamber, there is no further transfer of air from the cylinder to the chamber. At this moment therefore the combustion of a definitely known quantity of fuel and air takes place in the chamber.

The problem is now to determine the relation between the course of combustion in the chamber and measured quantities of CO_2 and O_2 and the known air content $L_Z = 0.00154$ m³ (15°, 1 atm.) of the chamber.

Let it be assumed that x kg of gas oil is completely burned at the crank angle α, whereby, from the 1 kg carbon content, the portion φ is separated in the form of soot; and that the portion β of the combustion gases flows from the chamber to the cylinder. The weight of the burned gas oil is then

$$x = \frac{4\ L_Z}{24.4\ h} \cdot \frac{0.21 - CO_2 - O_2}{1 - CO_2 - O_2}$$

and the portion

$$\varphi = 1 - 0.79 \frac{3\ h}{C} \frac{CO_2}{0.21 - CO_2 - O_2}.$$

The weight of the separated carbon is

$$C = 0.866\ \varphi\ x\ \text{kg}$$

From Figure 9 it is evident that the combustion in the chamber is completed in the time $Z_V = 0.0457$ s. During this time the quantity of fuel $B_{total} - B_z = 0.33$ g further entering through the nozzle, together with the separated carbon, is atomized by the flow into the cylinder and there consumed.

Since the fuel jet has only a small kinetic energy in the ignition-chamber engine, this energy must be increased by the blast from the chamber, so that the main combustion in the cylinder will be quick and complete. The kinetic energy of the d G_1'' kg of the combustion gases, flowing from the chamber into the cylinder at the velocity of w_1 m/s, is

$$dE'' = d\ G_1'' \frac{w_1^2}{2\ g} .$$

If p_1, as before, denotes the pressure in the cylinder at the time z or at the crank angle α, p_2 and T_2 the pressure and temperature in the chamber, then the kinetic energy of the blast from the chamber is

$$E'' = \frac{\mu\ f}{2\ g\ 6\ n\ R} \int_{\alpha_z}^{\alpha} \frac{P_1\ w_1^3}{T_2 \left(\frac{p_1}{p_2}\right)^{\frac{m-1}{m}}}\ d\ \alpha.$$

for the crank angle $\alpha - \alpha_z$. If it is desired to take into consideration the injected quantity of gas oil $B_{total} - B_z$ in addition to the gas weight G_1'', the energy of the chamber is increased only by the slight amount

$$E_b = \frac{B_{total} - B_z}{\alpha - \alpha_z} \int_{\alpha_z}^{\alpha} \frac{w_b^2}{2\,g} d\alpha \text{ mkg}$$

The main combustion causes a pressure increase in the cylinder, which affects the pressure $\Delta p = p_2 - p_1$ between the chamber and the cylinder, since it reduces Δp and consequently, w_1, and increases the duration of the blast from the chamber. The time interval z_v, in which the complete combustion of the chamber contents takes place, happens to coincide exactly with the blowing-off time of the chamber, since $\Delta p = 0$ at about 90° behind the crank dead center.

The composition of the water-containing combustion gases blowing out of the ignition chamber is changed by the progressive combustion in the chamber. At the crank angle α, let a_1 represent the carbon-dioxide content, a_2 the water vapor, a_3 the oxygen, and a_4 the nitrogen. The quantities a_1, a_2, a_3, and a_4 are known, since the quantity of gas oil burned $x = f(\alpha)$ has already been determined for the chamber.

The progress of the main combustion in the cylinder can now be described. When the piston is at the extreme dead center ($\alpha = 0^\circ$), G_{10} kg of air is enclosed in the cylinder, in which

the slight combustion residues can be regarded as part of the air. Up to the crank angle α_z° (ignition point) G_{2_z} kg of air has been expelled from the cylinder into the chamber. Since the direction of flow is reversed at the angle α_z° G_1'' kg of gas flows from the chamber into the cylinder between the angles α_z° and α°. Simultaneously, B_α kg of fuel is injected into the cylinder and C_α kg of unburned carbon is introduced from the chamber. At the crank angle α° the cylinder charge is therefore

$$G_{1_\alpha} = G_{1_0} - G_{2_z} + G_1'' + B_\alpha + C_\alpha \text{ kg}$$

When we consider that the ignition-chamber gases consist of CO_2, H_2O, O_2, and N_2, corresponding to $a_1 + a_2 + a_3 + a_4 = 1$, that the portion x of the fuel B injected into the cylinder and the carbon C are burned in the cylinder, we finally arrive at the formula

$$x = \frac{\left[\frac{0.231(G_{1_0}-G_{2_z})+a_3 G_1''}{32} + \frac{0.769(G_{1_0}-G_{2_z})+a_4 G_1''}{28}\right] CO_2 - \left(\frac{a_1 G_1''}{44} + \frac{C_\alpha}{12}\right)(1-CO_2)}{\frac{B_\alpha}{12}(c + 3\,h\,CO_2)},$$

in which CO_2 represents the carbon-dioxide content of the gas sample taken from the cylinder at the crank angle α with the aid of the controlled valve. The quantity of fuel actually burned in the cylinder at the crank angle α is then

$$B = x\ B_\alpha\ \text{kg}$$

The mixture of the fuel charge B_α and the carbon C_α with the air present in the cylinder before its combustion is denoted by the excess-air coefficient, which decreases with the progress of the fuel injection from the value $\lambda_{-50} = \infty$ down to the limiting value λ_{+310} at the end of the injection. Of the fuel quantity $B_{total} = 0.416$ g injected during each working cycle, the portion $B_z = 0.086$g is burned in the chamber and the remainder $B_{total} - B_z = 0.33$ g is burned in the cylinder. In Figure 10 the quantities involved in the main combustion in the cylinder are plotted against the crank angle α.

The course of the main combustion in the cylinder is thus fixed in all its details. The quantitative determinations explain the ignition-chamber processes in detail. The investigations show the fundamental influences exerted by the flow phenomena in the cylinder and ignition chamber on the functioning of the engine.

The Experimental Results and Their Significance for the Ignition-Chamber Process

An essential characteristic of the working process of the engine is that the fuel injected during each cycle is divided by the ignition in the chamber into two distinct parts. The first part $B_z = 0.086$ g is burned in the chamber with a considerable excess of air (46%) and causes a single preliminary explosion,

which effects the atomization of the second fuel portion $B_{total} - B_z = 0.33$ g in the cylinder. The atomization of the first fuel portion is effected by the action of the high air velocity in the connecting passage in its back flow into the chamber. The high chamber temperature and the exceptionally strong turbulence in the chamber greatly assist the incipient combustion.

The danger of carbonizing the chamber and nozzle, which is feared by all constructors, is effectively met by the presence of a large excess of air in the chamber. Since the very swift air flow simultaneously produces a very fine atomization of the fuel in the chamber, all the essentials (large area of the injected fuel, high temperature and high partial pressure of the oxygen in the chamber) are present for producing rapid oxidation of the fuel with rapid disintegration of the oxygen compounds formed and for promoting the ignition.

The immediate ignition of the fuel as it leaves the nozzle is prevented by the shortness of the distance from the nozzle to the mouth of the passage due to the flat shape of the chamber and by the fact that the fuel is injected into the passage as a solid jet. The short time and small surface prevent the ignition and any change in the fuel jet during the passage of the fuel through the chamber. Since, in the experimental angine, the distance is 11 mm and the jet velocity is 158 m/s, the jet traverses the chamber in 7×10^{-5} s, a period about 100 times

as small as required for the ignition of atomized fuel.

Another characteristic of this engine is the thorough water-cooling of the ignition chamber and connecting passage. The temperature measurements (Fig. 6) show that the inner-wall temperatures are low and that they nowhere exceed the usual wall temperatures of the water-cooled walls of ordinary internal combustion engines. The ignition is not affected by the wall temperatures either in the ignition chamber or in the passage.

The calorimetrically determined heat loss of the chamber and passage and the temperature measurements enable us to calculate the mean coefficient of heat transmission from the gases to the walls. For the ignition chamber this is $\alpha_i = 183$ k cal/m² h °C. The mean coefficient of heat transmission for the chamber according to Nüsselt's formula (Forschungsheft No. 264) is

$$\alpha_i = \frac{\int_0^{720} \alpha \, d \, \varphi}{720} = 181 \text{ k cal/m}^2 \text{ h } ^\circ\text{C}$$

This value agrees remarkably well with the measured value. The heat lost by radiation assumes some importance only at the maximum temperature, when it amounts to $\alpha_s = 44$ k cal/m² h °C. The mean coefficient of heat transmission by radiation is $\alpha_{s_m} =$ 9 k cal/m² h °C, so that the heat lost by the ignition chamber through radiation is only 5% of the total heat loss.

The effect of the heat loss from the cooling of the ignition chamber is generally much overestimated. It follows from the

heat balance that only 3.6% of the heat content of the fuel is lost by the cooling of the ignition chamber, while 38.8% is abstracted from the cylinder by the cooling water. Here again it has been found that the immediate heat loss through the walls plays only a subordinate role, as long as the energy conversion proceeds very rapidly.

The novelty is that the combustion in the ignition chamber takes place with an excess of air. During the introduction of the fuel $B_z = 0.086$ g up to the time of the ignition in the chamber, the coefficient of air excess decreases from $\lambda = \infty$ to $\lambda = 1.46$. The removal of the combustion products from the chamber is facilitated by the flow. Moreover, the exhaust and suction strokes of the engine piston exert a suction force on the chamber gases, the weight of which remaining in the chamber is small, due to the high temperature. The result of these circumstances is that the charging of the chamber with fresh air during every working cycle proceeds smoothly and produces the same mixture ratio of fuel and air and consequent uniform auxiliary explosions.

Naturally the combustion in the chamber requires a certain amount of time, if it is to be complete. This period, however, does not exceed the time which would otherwise be required under like conditions. As already mentioned, the time required for the complete combustion of the chamber contents was found to be $Z_V = 0.045$ second.

Notwithstanding the established complete combustion in the chamber (no CO and H_2 being discoverable in the chamber gases), the CO_2 content remained below its maximum value. This can only signify that the time was insufficient for the complete combustion of all the carbon compounds in the chemically non-homogeneous gas oil. As shown by all the observations of the combustion phenomena, the combustion is never complete with the ignition and first pressure rise. The carbon compounds, with their slower oxidation, require a longer time to absorb the heat needed to cause their disintegration and for the succeeding conversion into CO_2 and H_2O, thus resulting in an after-combustion. In the calculation this condition was satisfied by the assumption, on the basis of observations, that the combustion takes place with the separation of carbon. This circumstance is not necessarily unfavorable for the progress of the combustion, since glowing particles of carbon are excellent catalyzers and consequently carry the combustion very effectively from the ignition chamber into the cylinder.

With the same injection period, it is possible, by means of the preinjection angle, to regulate the amount of fuel desired for the chamber and thereby the mixture ratio of the fuel and air, as also the explosion in the chamber, and consequently the expulsive force. There always remains, however, an excess of air, which is important for the ignition and combustion in the chamber.

While, in other ignition-chamber processes, the first portion of the fuel charge is burned with a considerable lack of air and consequently with incomplete pressure rise in the ignition chamber and the endeavor is made to heat strongly, in the absence of oxygen, the main portion of the charge just before its injection into the ignition chamber, thus splitting up the fuel molecules, if possible, the first portion of the fuel charge here comes immediately in contact with excess oxygen in a state of maximum turbulence. Thus the preliminary explosion in the chamber can be fully controlled. The utilization of the high velocity of the air flow in the chamber for atomizing the injected fuel makes it possible, moreover, to work with a lower pump pressure, since the disintegration of the jet is not caused by high pump pressure, as in ordinary airless-injection engines, but by turbulence.

Even the expelled ignition-chamber gases still contains a large quantity of free oxygen, which immediately goes to aid the main combustion in the cylinder. Consequently, the latter combustion takes less time (Fig. 10).

The fact that the ignition conditions in the chamber are extremely favorable for the chosen working cycle, explains why the engine, without structural changes, can quickly and surely burn even heavy oils having a high ignition point.

The kinetic energy of the air jet entering the chamber serves as a criterion for the strength of the turbulence. For

the compression stroke this energy is 8.3 mkg or 0.46 mkg/hp. For comparison, the values are quoted which I previously reported (V.D.I., 1923, p. 755) in connection with the turbulence produced by the injection air of a 50 hp engine, where the kinetic energy was found to be 0.3 mkg/hp. It is evident that the turbulence is 50% greater in the airless-injection engine under consideration. Based on the total amount of fuel injected, the effect of the air jet on the atomization seems to be more favorable for the ignition-chamber engine under consideration. The turbulence in the ignition chamber and the expulsive power of the chamber together produce a kinetic energy of 14.3 mkg. The injected fuel weighs 0.416×10^{-3} kg. Hence $E = 34,300$ mkg for 1 kg of fuel, while only 8970 mkg/kg is available for the air-injection engine.

Since, in both cases, the fuel particles during the atomization are preferably brought into contact with pure air, it is obvious that the fuel in the ignition-chamber engine, up to the beginning of the combustion, is not more poorly prepared in any case than in the air-injection engine. The ignition should be even better in the ignition-chamber engine on account of the elimination of the cooling effect of the injection air.

The fact that the relative velocities of the fuel and air in the investigated ignition-chamber engine are not smaller than in the original Diesel engine favors the good atomization of the fuel. It is essential, however, for the fuel to be in-

jected into the chamber at the exact instant that $dE/d\alpha$ reaches its maximum value.

The course of the main combustion in the engine and hence the real working process depend essentially on the blow-off energy of the chamber, calculated from the moment of the ignition up to the pressure equalization between the chamber and cylinder. This energy is

$$E'' = \frac{\mu\ f}{2\ g\ 6\ n\ R} \int_{\alpha_z}^{\alpha} \frac{p_1\ w_1{}^3}{T_2 \left|\frac{p_1}{p_2}\right|^{\frac{m-1}{m}}}\ d\ \alpha\ \text{mkg}$$

At constant r.p.m. the chamber pressure p_2 is a function of the air excess with which the portion of the injected fuel introduced into the ignition chamber burns. The smaller this portion is, just so much smaller the heat content of the chamber charge and the chamber pressure p_2 are. Then, however, the portion of the fuel left for the cylinder and the cylinder pressure p_1 produced by the combustion are greater. This reduces, however, the exhaust velocity w_1 of the burned gases, since the latter depends chiefly on the pressure difference $p_2 - p_1$.

In order to obtain the maximum blow-off energy of the chamber, the injection ratios (i.e., the division, for each working cycle, of all the injected fuel B_{total} into the chamber portion B_z and the cylinder portion $B_{total} - B_z$ kg) are so chosen that p_2, in comparison with p_1, and consequently w_1 are as great

as possible. There is then the further advantage of a short blow-off time whereby the main fuel charge is injected into the cylinder near or slightly after the dead center, which promotes quick combustion at the beginning of the expansion stroke. A low chamber temperature T_2 increases the weight of the escaping chamber gases and consequently the energy of the chamber, but militates, on the other hand, against the certainty of ignition in the chamber.

The energy of the escaping ignition-chamber gases varies greatly. It is evident that $d\, \Sigma\, E''/d\, \alpha$ has a decidedly maximum value near the dead center at the instant the fuel is injected into the cylinder. The mean value of the flow energy is here greatly exceeded, with a corresponding increase in the turbulence which greatly promotes the main combustion in the cylinder. As in the ignition chamber, also here in the cylinder the fuel injection coincides in time with the maximum turbulence, which may explain the good combustion properties of this engine.

The succession of the combustion phases can be easily determined for the ignition chamber. Since the beginning of the fuel delivery by the pump is at 17°, the beginning of the injection at 13°, and the ignition at 5° before the dead center and the maximum pressure in the chamber occurs at 5° after the dead center, the injection lag is 1.92×10^{-3} s and the ignition lag is 3.84×10^{-3} s, corresponding to $z = \Delta\alpha/6n$. The main combustion in the chamber is completed in 4.8×10^{-3} s.

That the ignition lag of about 1/250 s is rather large, despite the high temperature and great gas density in the chamber, is due to the fact that the fuel injected into the connecting passage, must first be driven back into the chamber before it finds the conditions for ignition. This requires, however, a crank angle of 8° or a time interval of 3.84×10^{-3} s.

No details can be given regarding the main combustion in the cylinder. It is certain, however, that the preparation of the fuel, up to the combustion, is not accomplished during its passage through the ignition chamber and the connecting passage, since both the distance and the time are much too small. The lagging of the combustion behind the injection clearly shows that the vaporization of the originally liquid fuel occurs in the cylinder. (Compare the $x\ B_\alpha$ curve with the B_α curve in Figure 10.)

The course of the main combustion in the cylinder of an ignition-chamber engine does not differ much from that in an air-injection engine, as I have previously established (Forschungsheft No. 245, Fig. 14). The combustion velocity $\frac{d(x\ B_\alpha)}{d_z}$ g/s can be calculated from the curve of the combustion in the cylinder by differentiation. From the beginning of the injection into the cylinder, the combustion velocity tends to approach very rapidly a maximum which evidently coincides with the instant of best atomization (maximum surface area of the fuel cloud) and maximum temperature. Then the injection turbulence

$d\,\Sigma\,E''/d\,\alpha$ in the chamber and consequently the atomization decreases rapidly. Moreover, due to the output of the engine, the temperature of the cylinder contents drops, whereby the combustion velocity decreases, the injection sometimes ending at a crank angle of 31° and the easily oxidizable gas-oil components burning up very rapidly.

All these phenomena demonstrate that the fuel designed for the cylinder ($B_{cycle} - B_z$) obtains the heat required for its preliminary conversion up to the combustion neither in the chamber nor in the passage but only in the cylinder.

The heat absorbed by the cooling water is greater in ignition-chamber engines of small power than in ordinary airless-injection engines of the same size. This, however, involves no detriment to the economical efficiency η_{te}, since the heat lost in the exhaust diminishes with increasing heat absorption by the cooling water. For the investigated engine, the ratio of the heat absorbed by the cooling water to that carried away by the exhaust was $q_k/q_z = 1.72$ (See "Heat Balance"). The ignition chamber participates in the cooling loss only in a subordinate degree. The cooling loss of the chamber is 8.6%, of the connecting passage 5.7%, while the remaining 85.7% belongs to the cylinder. It can be easily demonstrated that, during a four-stroke cycle, the heat is largely absorbed by the cylinder wall during the combustion, in which, as shown above, the turbulence developed in the cylinder by the ignition-chamber combustion

attains its maximum strength $(d\,\Sigma\,E''/d\,\alpha)_{max}$.

For a smaller combustion space, however, as in a low-powered engine, the effect of the turbulence velocity of the burning charge on the heat transmission to the corresponding wall areas of the combustion space is more noticeable than in large engines. This was demonstrated by experiments in 1925 with a 300 hp horizontal two-cylinder Körting engine. This engine had a stroke s of 850 mm, a cylinder diameter d of 495 mm, a cylinder volume V_h of 163.7 liters, and a nominal speed n of 160 r.p.m. The tests at full load gave the following heat balance.

	Cylinder I	Cylinder II
Useful work q_e	0.356	0.359
Frictional work q_ρ	0.051	0.052
Cooling water q_k	0.286	0.280
Exhaust gases q_z	0.268	0.301
Remainder q_r	0.039	0.008
Σq	1.000	1.000

On the average therefore, the heat absorbed by the cooling water equals that carried off by the exhaust gases, somewhat as in solid-injection engines of the same power. With an economical efficiency of $\eta_{t_e} = 36\%$, this engine shows that the ignition-chamber process, even for large cylinder outputs, is equivalent to pressure injection.

While the cooling loss for the 18 hp one-cylinder laborato-

ry engine was 42.4%, it dropped to 28.3% (only about 2/3 as much) for the 150 hp engine. For these engines the ratio of the surface area of the combustion space is $O : V = 150$ and 44.8, respectively. If we put, as a first approximation, the cooling losses directly proportional to this ratio and to the time, i.e., inversely proportional to the r.p.m., we get

$$\frac{Q_{k_{18\ hp}}}{Q_{k_{150\ hp}}} = 1.54,$$

that is, the same value as computed from the measured cooling losses

$$\frac{q_{k_{18\ hp}}}{q_{k_{150\ hp}}} = 1.50.$$

The greater heat consumption per horsepower of the laboratory engine is due, therefore, not to greater cooling losses, but to the relatively long injection time. On using another fuel-pump cam, the fuel consumption returns to a value approaching that for ignition-chamber engines of this size.

In my presentation of the subject, I have referred to only a few points, the elucidation of which seems to be the most important at the present time and for which the experiments afford sufficiently reliable data. The high-speed development, which now dominates engine construction, forces the dynamics of the combustion process into the foreground of consideration, for the shorter the available combustion time is (In high-speed engines

it is only a few thousandths of a second), the quicker must be the chemical reaction in order to attain high fuel efficiency.

The ignition-chamber process must be advantageous for high-speed Diesel engines, because quick heat transmission to the injected fuel can be effected by simple means with good atomization through high temperatures and great turbulence. The more quickly the beginning of the ignition is followed by the formation of gas from the disintegration of the molecules, just so much quicker the combustion will end and just so much less it will be affected by outward heat losses.

The development of the ignition-chamber engine to its present form has covered a long period of time. In its beginnings this development goes back to the work of Dr. Rudolf Diesel. I believe the top of this ascending curve has not yet been reached. As so often happens in technical matters, practical construction has gone ahead of scientific information. It is to be hoped, however, that the increasing insight into the nature of the working process of this engine will help, in the future, to increase the fuel efficiency even for this type of engine.

In the use of heavy oils with high ignition points and large carbon content, the ignition-chamber engine is superior to the airless-injection engine. Its reliability is beyond question even under difficult conditions.

Translation by Dwight M. Miner,
National Advisory Committee
for Aeronautics.

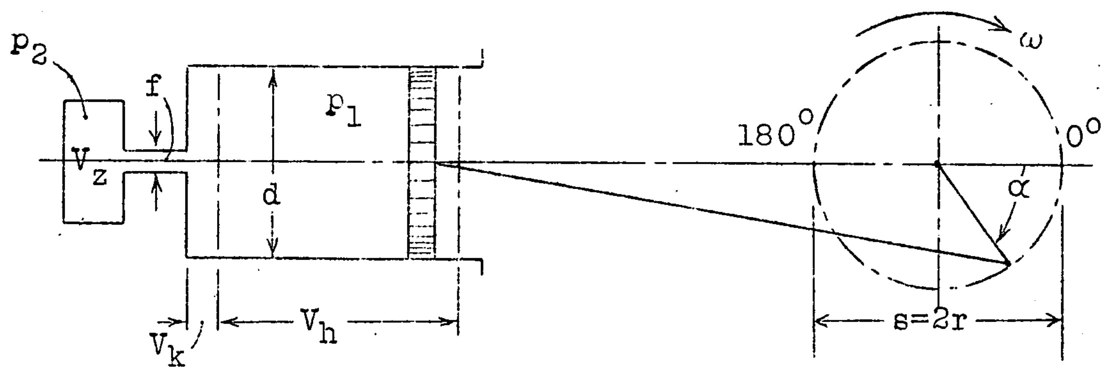
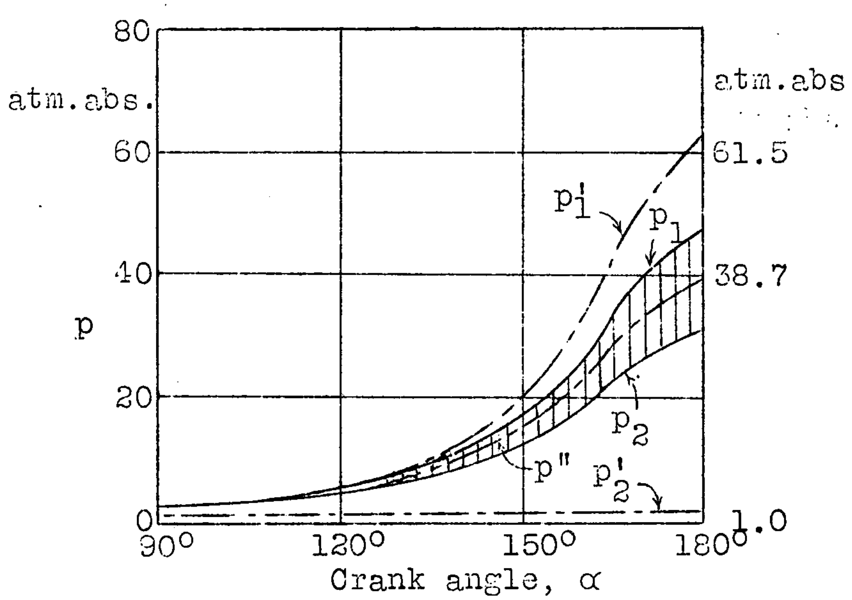


Fig. 1 Diagram of section through cylinder and ignition chamber.



p_1, pressure in cylinder
p_2, " " chamber
p_1', p_2', limiting pressure for $n=\infty$
p'', " " " $n=0$

Fig. 2 Pressure curves in cylinder and ignition chamber near end of compression stroke.

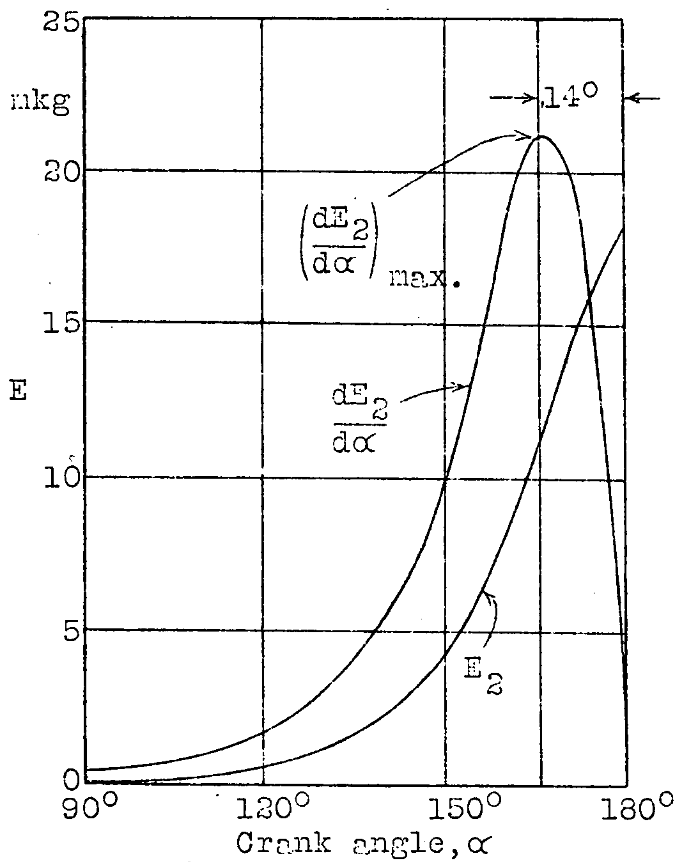


Fig. 4

G_2, weight of air expelled from cylinder into chamber.
w_2, velocity of flow.
E_2, kinetic energy of flow.
$\frac{dE_2}{d\alpha}$, curve of kinetic energy.

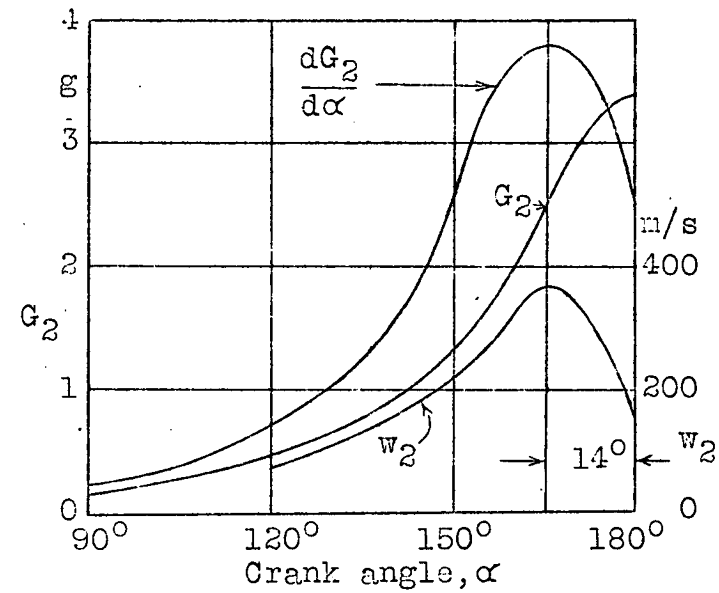


Fig. 3

Flow phenomena in chamber near end of compression stroke at constant engine speed.
(n=311 r.p.m.)

G_2, weight of air expelled from cylinder into chamber during compression.
$w_{2_{max.}}$, maximum velocity of flow during compression stroke.
E_2, kinetic energy of expelled air.
$\left(\frac{dE_2}{d\alpha}\right)_{max.}$, maximum turbulence in chamber.

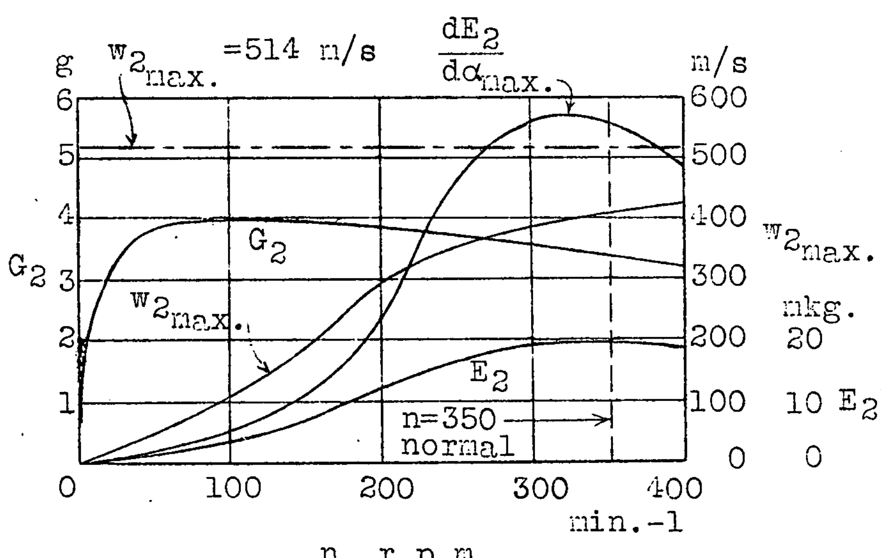


Fig. 5 Flow phenomena in chamber plotted against r.p.m.

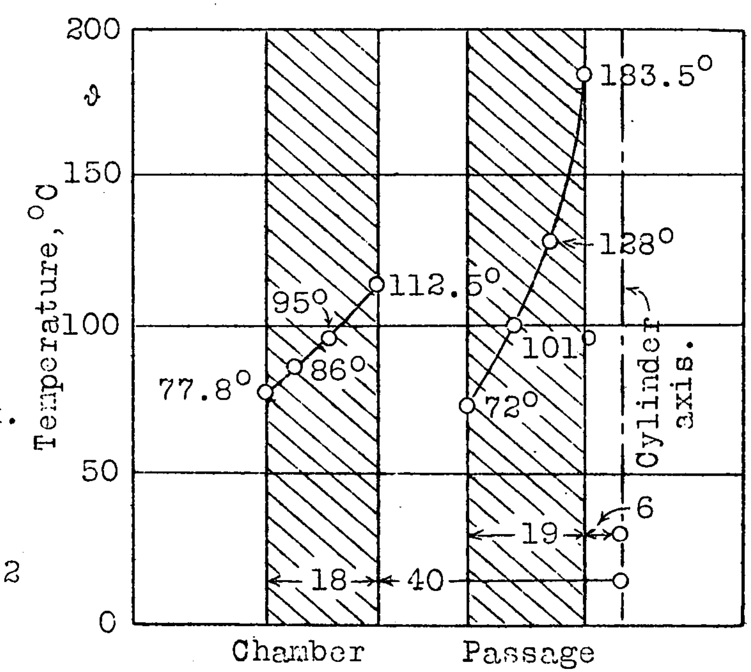
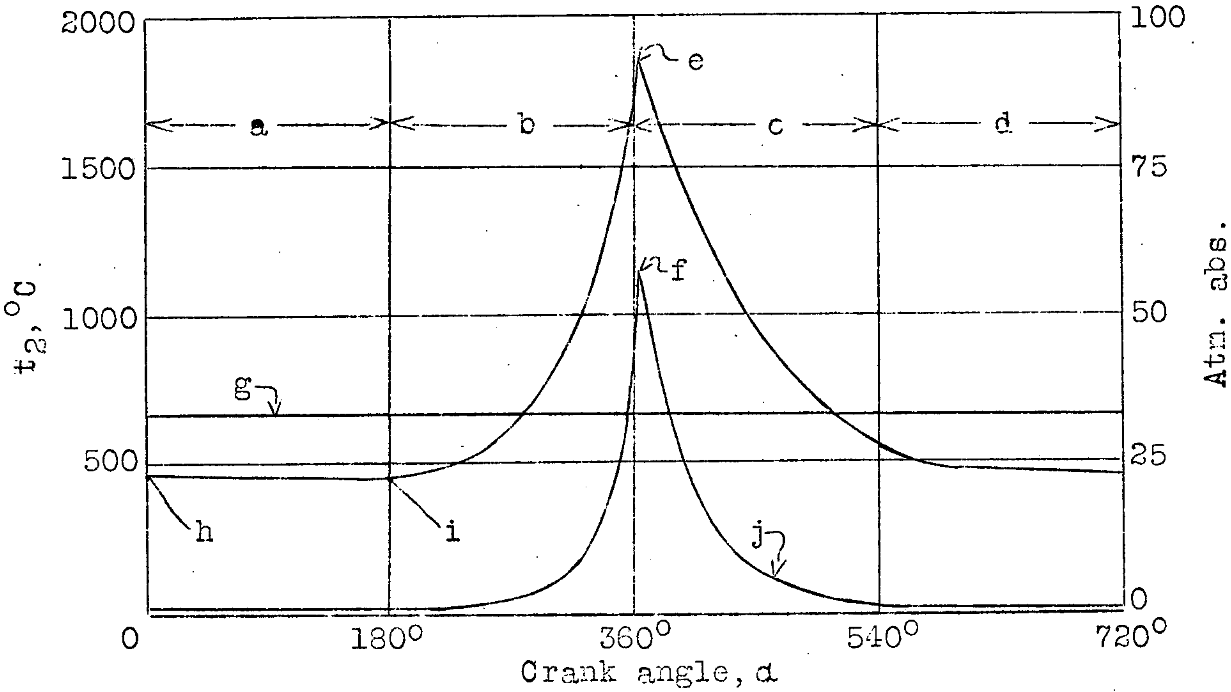


Fig.6 Temperature curves in walls of chamber and connecting passage.

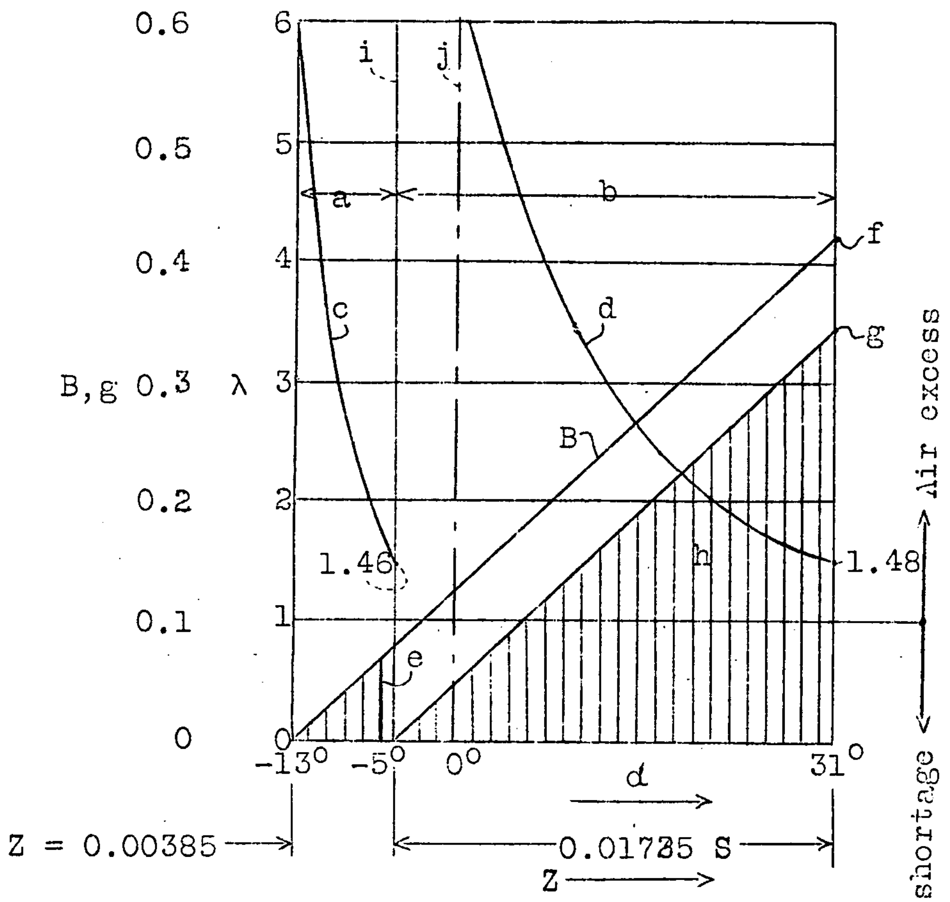


a, Suction
b, Compression
c, Expansion
d, Exhaust
e, $t_{2max.}$=1857°

f, $p_{2max.}$=57.1 Atm. abs.
g, t_{2m}=672°
h, t_{20}=477°
i, t_{21}=462°
j, p_2

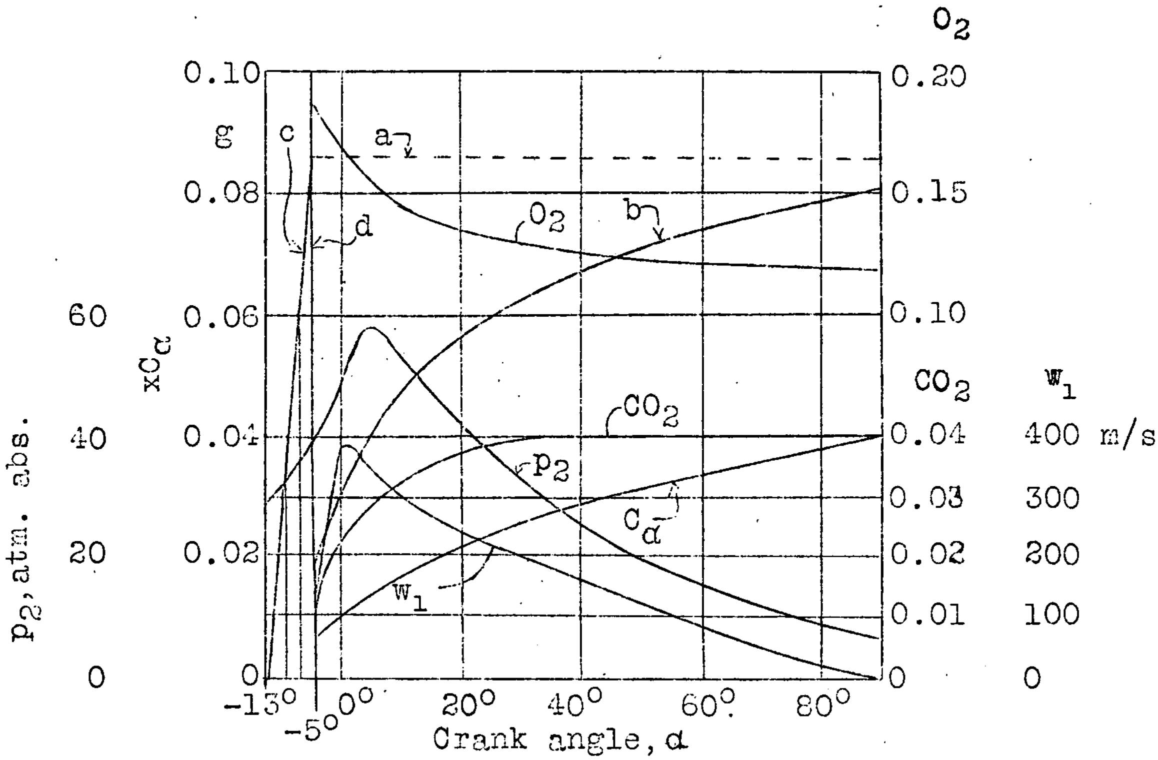
Fig.7 Pressure and temperature curves in chamber during four-stroke cycle.

B, Weight of fuel injected (in grams).
λ, Coefficient of air excess.



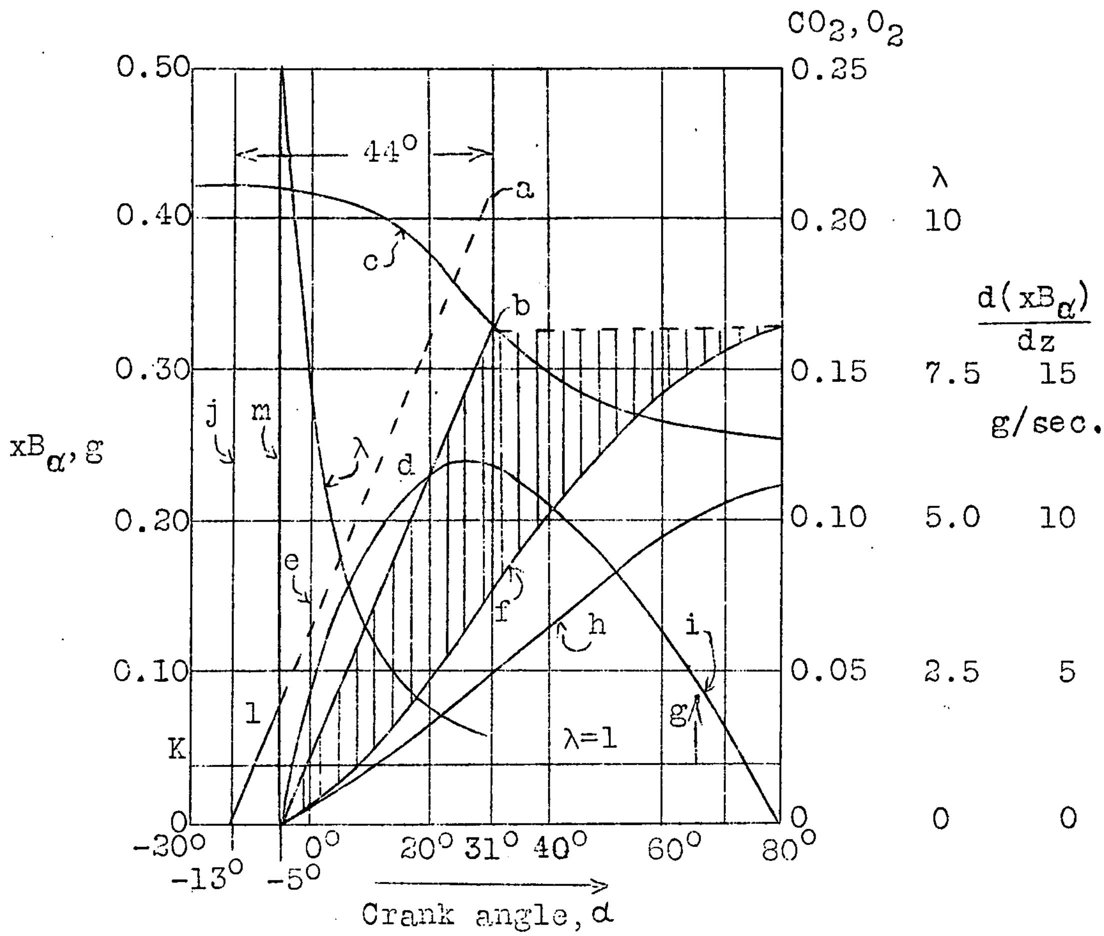
a, Injected into chamber
b, Injected into cylinder
c, $\lambda_{chamber}$
d, $\lambda_{cylinder}$
e, $B_{chamber}$ = 0.086 g
f, B_{total} = 0.416 g
g, $B_{cylinder}$ = 0.330 g
h, λ_{chem} = 1
i, Ignition
j, Dead center

Fig.8 Injection curves for chamber and cylinder.



a, B_Z=0.086 g B_z, Fuel injected
b, Burned, x X , Fuel burned
c, Injected C_α, Carbon liberated
d, Ignition
p_2, Pressure in chamber
w_1, Blow-off velocity from chamber to cylinder.
CO_2, CO_2 content (volumetric) of dry combustion gases.
O_2, Oxygen content (volumetric) " " " .

Fig.9 Combustion curves in chamber.



a, B_{total}=0.416 g
b, $B_{cylinder}$=0.330 g
c, O_2
d, Injected, B_α
e, Dead center
f, xB_α, burned
g, Air excess
B_α, Fuel injected into cylinder
xB_α, Fuel burned
$\frac{d(xB_\alpha)}{dz}$, Combustion velocity

h, CO_2
i, $\frac{d(xB_\alpha)}{dz}$
j, Needle valve opens
l, B=0.086 g
m, Ignition

CO_2, O_2: CO_2 and O content (volumetric) of dry combustion gases
λ, Coefficient of air excess

Fig.10 Combustion curves in cylinder.